Design and Analysis of Electro-Hydraulic Servo System for Speed Control of Hydraulic Motor

Dr. Ali Abdul Mohsin Hassan AL-Assady
Asst. prof. Dept. of Mech. Eng.
Email: dralicit@yahoo.com
Baghdad University

Mohammad Talib. Jassim AL-Khafaji
Dept. of Mech. Eng.
Email: mohammad_t.jassim @yahoo.com
Baghdad University

ABSTRACT

In this study, the electro-hydraulic servo system for speed control of fixed displacement hydraulic motor using proportional valve and (PID) controller is investigated theoretically ,experimentally and simulation . The theoretical part includes the derivation of the nonlinear mathematical model equation of (valve – motor) combination system and the derivation of the transfer function for the complete hydraulic system , the stability test of the system during the operation through the transfer function using MATLAB package V7.1 have been done. An experimental part includes design and built hydraulic test rig and simple PID controller .The best PID gains have been calculated experimentally and simulation, speed control performance tests for the system at different thermal conditions for hydraulic oil have been done , Simulation analysis for (EHSS) using Automation Studio package V5.2 have been done . Comparison was made between experimental work and simulation work .The experimental results show good performance for (EHSS) using simple (PID) controller at hydraulic oil temperature around (60 – 70 ℃) and good speed response and performance for hydraulic motor with constant rotation speed (700) rpm with different load disturbance applied on the hydraulic motor .

KEYWORDS: **Servo, PID controller, Proportional valve, Hydraulic motor, Automation Studio (AS)**

تصميم وتحليل منظومة الكترو- هيدروليكية مغلقة للسيطرة والتحكم في سرعة محرك هيدروليكي

جاسم الخفاجي طالب محمد

أ.م.د علي عبد المحسن حسن الاسدي

الخلاصة

يتناول هذا البحث دراسة نظرية وعملية وبطريقة المحاكاة لمنظومة الكترو-هيدروليكية للتحكم في سرعة دوران محرك هيدروليكي ثابت السعة باتجاة واحد بأستخدام صمام تناسبي .الجانب النظري يتضمن أستقاق المعادلات اللا خطية للنموذج الرياضي لنظام مجموعة (الصمام – محرك) واشتقاق معادلة دالة الانتقال (transfer function) للنظام الهيدروليكي الكامل .تم أجراء فحص الاسقرارية اثناء التشغيل بواسطة معادلة دالة الانتقال وبستخدام برنامج (MATLAB package V7.1) .الجانب العملي يتضمن تصميم وبناء جهاز الاختبارالهيدروليكي وتصميم وبناء دائرة التحكم بسيطة (PID controller) . تم ايجاد افضل قيم للثوابت دائرة التحكم عمليا وبأستخدام برنامج المحاكاة .تم اجراء فحص أداء التحكم بالسرعة للمنظومة في ظروف حرارية مختلفة لزيت الهيدروليك. جانب الدراسة الخاص بالمحاكات التشبيهية للمنظومة الهيدروليكية تم بستخدام برنامج المحاكاة الخاص بالمنظومات الهيدروليكية المسمى (Automation Studio package V5.2) .المقارنة تمت بين النتائج العملية ونتائج المحاكات للبرنامج .النتائج العملية اظهرت اداء جيد للمنظومة بأستخدام زيت الهيدروليك عند درجة حرارة تتراوح بين 60 الى 70 درجة سيليزية واستخدام دائرة تحكم بسيطة (PID controller).تم الحصول على نتائج متقاربة للنتائج العملية بأستخدام المحاكاة للمنظومة .استجابة واداء جيد للتحكم بسرعة الموتور الهيدروليكي واستقرارها عند سرعة دورانية ثابتة (700) (دورة / دقيقة) تحت تسليط احمال مختلفة على المحرك الهيدروليكي.

الكلمات الرئيسية :- النظام المغلق ، منظم تناسبي تكاملي اشتقاقي ،صمام تناسبي ،محرك هيدروليكي ،برنامج المحاكات الدوائر الهيدروليكية

INTRODUCTION

Electro-hydraulic servo system (EHSS) is widely used in many industrial applications and mobile systems because of their high power-to-weight ratio, high stiffness, fast response, self cooling, good positioning capabilities, etc. Either of two basic methods is used for electro hydraulic servo system for speed control of a hydraulic motor.

First, a variable displacement pump controls flow to the motor. Second, a servo or proportional valves Second method is a closed loop speed control system the actuating error signal which is the difference between the input signal and the feedback signal which is fed to the controlled so as to reduce the error and bring the output of the system to a desired value .This work is focused on study fixed displacement pump-Fixed displacement motor by using proportional valve PV and PID controller. The closed- loop of speed control as shown in **fig. 1** Vladimir (2006).

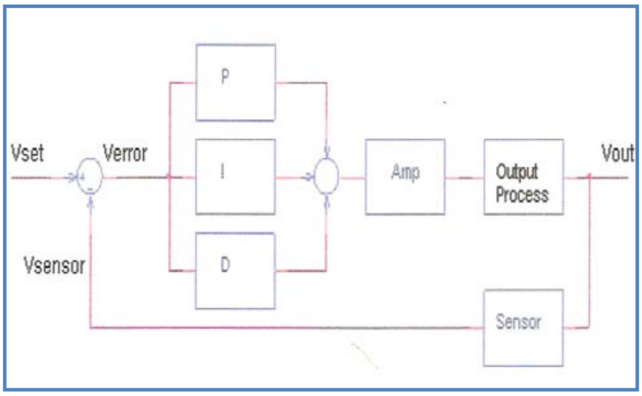

Fig (1): Schematic diagram of the Servo control system with PID controller

The error signal is control signal which controlled using (PID) controller (proportional integral derivative) controller .this type of controller is still the most popular ones in the processing industries for speed control.

There are several circuits for speed control system like meter – in flow controls, meter – out flow controls, bleed – out flow controls and variable volume pump .the meter – in/out flow controls as shown in **fig. 2** (Web site 1).

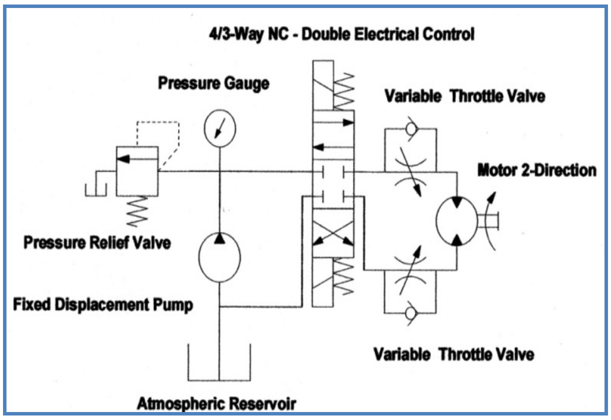

Fig (2): Motor circuit with meter – in / out flow controls

Much research work have been accomplished in the area of the performance investigation of(EHSS) for speed control with various combinations of pumps and motors.

Ambuel (1993) introduced an experimental and analytical study dealt fuzzy – PI control applied to speed control of fixed displacement hydraulic motor. They found that the performance characteristics of a hydraulic motor under conventional (PI) control could be improved by implementing a fuzzy– PI controller design.

Yasukazu (2000) **,** introduced simulations and experimental study dealt with new type for rotational speed control of hydraulic motor with large inertia loud by using sliding mode control**.** the result of theoretical and experimental study of sliding mode control show many benefits such as setting time and low sensitivity to disturbances and system parameter variations in the (HS).

Aichao et al (2008) introduced simulation and analytical study dealt with the speed regulation fundamental principle of complete system fixed displacement pump- variable displacement motor. and use integral control . The AME Sim simulation model of the system was Built, analyzing the influences of relevant factors on constant speed output control, The simulation results provides theoretical basis for designing the constant speed output of the hydraulic motor under change of many factors and give good response of the system output .

Dasgupta et al (2011) introduced modeling and simulation study dealt with comprehensive model of closed- loop servo valve controlled hydro motor drive system has been made using (Bond graph simulation technique) . The dynamic performance of the complete system has been studied with respect to the variation of the parameters of the (PI) controller that drives the servo valve, They have also studied the effects of the variation of torque motor parameters on the servo valve performance using MATLAB simulik environment.

Hossam et al(2011) introduced an experimental and theoretical study of (EHSS) of speed control of hydraulic motor .The performance of the model-based control system depends strongly on the accuracy of the process model used. Least squares support vector machines method (LS-SVM) is powerful method for modeling nonlinear system results show good performance over wide range of operating conditions and load disturbances.

In the present work, the objective is to investigate experimentally speed control system for the hydraulic motor using fixed displacement pump and proportional valve with a simple (PID) controller under different range of hydraulic temperature and simulation analysis for the system with Automation Studio (AS) package V5.2.

THEORETICAL ANALYSIS

The modeling of the system has been performed depend on assumptions (Vladimir 2006):-

- A constant pressure source
- Fluid inertia is neglected.
- No flow reversal or cavitations
- Return line pressure (P_o) is neglected
- Orifices are matched and symmetrical

MATHEMATICAL DESCRIPTION OF THE PV AND HYDRAULIC MOTOR

The nonlinear dynamic model of (EHSS) shown in **Fig. 3**, which consists of a hydraulic motor controlled with proportional valve. (Merritt1967)

Equations of flow through the PV are derived from the application of flow continuity through the orifices of proportional valve and defined by the following terms:-

$$Q_1 = C_d A \sqrt{\frac{2}{\rho}(P_s - P_1)} \qquad (1)$$

$$Q_1 = C_d A \sqrt{\frac{2}{\rho}(P_s - P_2)} \qquad (2)$$

$$Q_2 = C_d A \sqrt{\frac{2}{\rho}(P_2 - P_o)} \qquad (3)$$

$$Q_2 = C_d A \sqrt{\frac{2}{\rho}(P_1 - P_o)} \qquad (4)$$

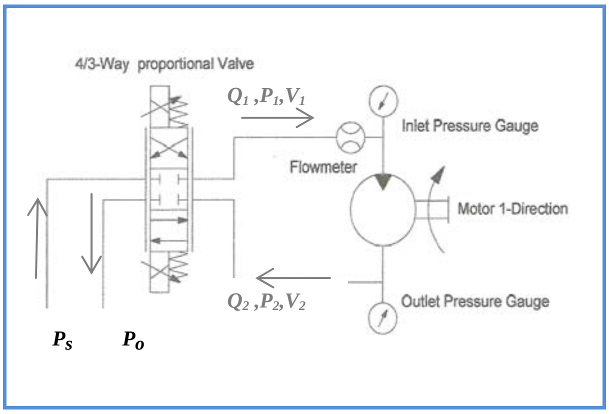

Fig (3): Schematic diagram of (valve – motor) combination system

If internal and external leakages are neglected, hydraulic pressure behavior for a compressible fluid volumes is given by the differential equations (1,2,3,4) :-

$$Q_L = C_d \,|A|\, \frac{x_v}{|x_v|} \sqrt{\frac{1}{\rho}\left(P_s - \frac{x_v}{|x_v|} P_L\right)} \qquad (5)$$

Where Q_L is the load flow threw the motor and P_L is the loud pressure. Linearized flow equation, which describes dynamic behavior of the proportional valve and around an operating point, is as follows:-

$$\Delta Q_L = K_q \,\Delta X_v - K_c \,\Delta P_L \qquad (6)$$

Where Kq is the flow gain of the PV and K_c is the pressure flow gain of the valve. K_q and K_c are given by:-

$$K_q = C_d \pi \, d_{spool} \sqrt{\frac{1}{\rho}(P_s - P_L)} \qquad (7)$$

$$K_c = \frac{C_d \pi \, d_{spool} \; X_v}{2\sqrt{\frac{1}{\rho}(P_s - P_L)}} \qquad (8)$$

By assuming:-

$$Q_L = \frac{Q_1 + Q_2}{2} \tag{9}$$

$$P_L = P_1 - P_2 \tag{10}$$

The continuity equations for each motor chamber same equations. (Merritt 1967):-

$$Q_1 - C_{im}(P_1 - P_2) - C_{em}P_1 = D_m\,\omega_m + \frac{V_1}{\beta}\frac{dP_1}{dt} \tag{11}$$

$$C_{im}(P_1 - P_2) - C_{em}P_2 - Q_{mo} = -D_m\,\omega_m + \frac{V_2}{\beta}\frac{dP_2}{dt} \tag{12}$$

From equations 9,10,11,12

$$Q_L = D_m\,\omega_m + C_{tm}\,P_L + \frac{V_t}{2\beta}P_L s \tag{13}$$

Where $V_1 + V_2 = V_t$

The torque equation for motor is given by :-

$$T_g = (P_1 - P_2)D_m = J_m\,\omega_m s + B_m\,\omega_m + T_{fm} + T_m + T_s \tag{14}$$

From equations 6, 13 and 14, over all TF for the system is given by :-

$$\omega_m = \frac{\frac{K_q}{D_m}X_v - \frac{TK_{ce}}{D_m^2}\left(1 + \frac{V_t}{2\beta K_{ce}}s\right)}{\frac{J_m V_t}{2\beta D_m^2}s^2 + \left(\frac{J_m K_{ce}}{D_m^2} + \frac{B_m V_t}{2\beta D_m^2}\right)s + \frac{B_m K_{ce}}{D_m^2} + 1} \tag{15}$$

Or two TF for the system (no load speed) and and (speed drop due to load) respectively:-

$$\frac{\omega_m}{x_v} = \frac{\frac{K_q}{D_m}}{\left(\frac{s^2}{\omega_h^2} + \frac{2\delta_h}{\omega_h}s + 1\right)} \tag{16}$$

$$\frac{\omega_m}{T} = -\frac{\frac{K_{ce}}{D_m^2}\left(1 + \frac{V_t}{2\beta K_{ce}}s\right)}{\left(\frac{s^2}{\omega_h^2} + \frac{2\delta_h}{\omega_h}s + 1\right)} \tag{17}$$

Where national frequency and damping ratio are given by:-

$$\omega_h = \sqrt{\frac{2\beta D_m^2}{V_t J_m}} \tag{18}$$

$$\frac{B_m}{2D_m}\sqrt{\frac{V_t}{\beta J_m}} + \delta_h = \frac{K_{ce}}{D_m}\sqrt{\frac{\beta J_m}{V_t}} \tag{19}$$

ANALYSIS OF (PID) CONTROLLER SYSTEM

(PID) controllers are simple controller for servo system, as shown in **fig. 4:-**

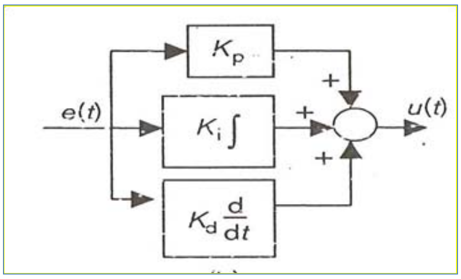

Fig (4): (PID) controller block diagram

A typical (PID) controller has following transfer function form is:-

$$u(t) = K_p\,e(t) + K_i\int_0^t e(t)dt + K_d\frac{d\,e(t)}{dt} \tag{20}$$

(PID) controller could be determined by many methods which lead to regulate the (PID) controller for example trial and error method or Ziegler and Nichols Jacqueline (2008)

Stability Test Using MATLAB Package V7.1

This analysis include the system stability analysis in accordance to the TF . The equations 19,20 .**fig. 5** is block diagram using MATLAB package V 7.1 .

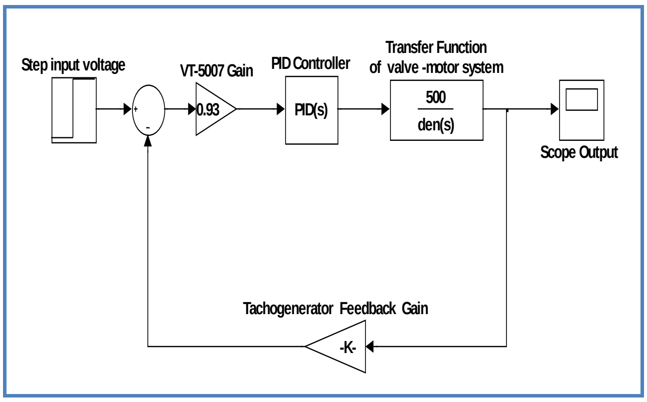

Fig (5): Block diagram of hydraulic control system with (PID) controller.

EXPERIMENTAL WORK

Design and Build the Test Rig

The test rig for this work is shown schematically in **fig. 6** and photographically in **fig. 7.**

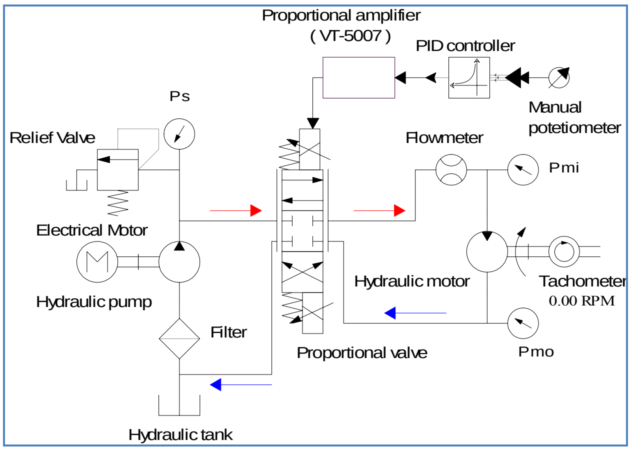

Fig (6): Schematic diagram of (EHSS) for speed control

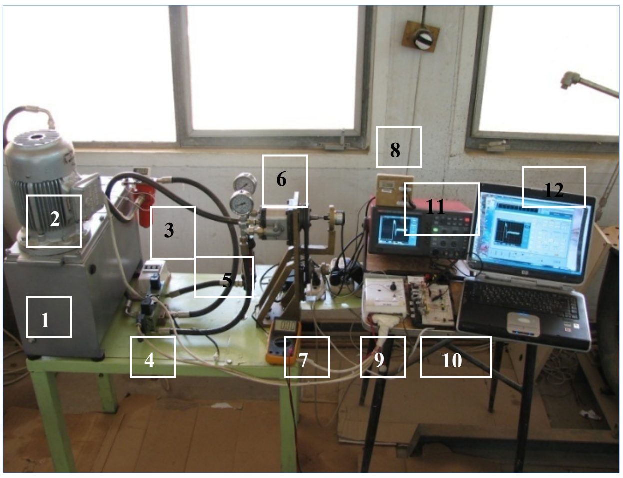

Fig (7): (EHSS) for speed control test rig

This test rig structure have many parts according to **fig. 6** , The hydraulic power unit (1) utilizes a gear pump (Duplomatic -11 cm^3/rev) to deliver the fluid to the system at a maximum rate of 14 l/min and maximum nominal pressure of 75 bar. The oil pump is driven by an electrical motor (2) (1.5 kW at 1400 rpm). high pressure filter (3) is used to protected the hydraulic system. The electro-hydraulic proportional valve manufactured by Bosch Rexroth (4WRAE 6 E16-10/24Z4/M) (4) directly controlled by proportional amplifier (VT-5007) (9) threw simple (PID) controller (10) ,the hydraulic motor (6) is fixed displacement external gear motor (Cassapa -11 cm^3/rev) with tow direction of rotation and leakage port, the rotation speed of the hydraulic motor transmitting to the sliding pulley which coupling with feedback tachogenerator sensor (7)using (V-belt), two pressure gauges (new –Tec) have been used in the hydraulic system and flow meter ,using digital tachometer (Autonics) (8) to record the rotation speed of hydraulic motor ,the output speed response of hydraulic motor is recorded by using digital oscilloscope (11) and save data on PC (12).

DESIGN AND BUILD (PID) CONTROLLER

This simple (PID) controller designed and built experimentally and as shown schematically in **fig. 8** and photographically in **fig. 9**, this controller is design and build experimentally by using 7 omp. amplifier type LM 741 , 5 potentiometers 250,100 KΩ , Capacitors 1 ,10 μF, -12 to 12 DC Power supply , Breadboard and jumper wire .(Thrower . et al 1998).

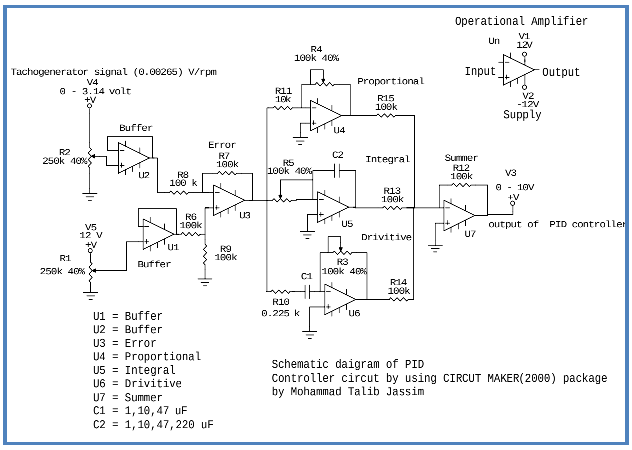

Fig (8): Schematic diagram of (PID) Controller

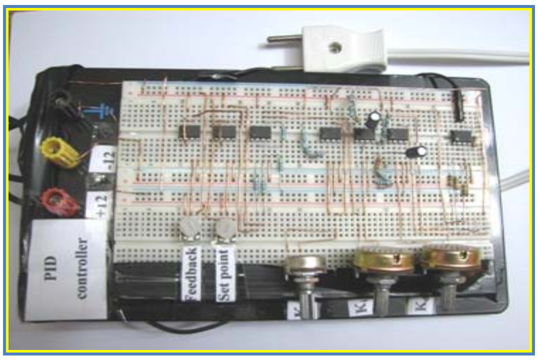

Fig (9) : Simple (PID) controller circuit

EXPERIMENTAL TESTING

Several comprehensive tests have been conducted on the (EHSS) for speed control test rig to investigate the hydraulic system and controller design performance and compare it against several studies. The first test was to calculate the total friction torque of the hydraulic motor and transmission system with V-belt . Second test was to verify the (PV) and hydraulic motor characteristic input voltage with rotation speed and flow rate and calculate the gain of the proportional amplifier experimentally. third test was open - loop system test by disconnected feedback signal of tachogenerator to check the stability and the response of the system .forth test was closed- loop system by connecting feedback signal of tachogenerator and (PID) controller test to evaluated the best (PID) gains for this system .last test was the speed control test at **(45 $\pm$5)** °C and (**65 $\pm$5**) °C temperature of hydraulic oil and using throttle valve working as variable load applied on the hydraulic motor by change the outlet flow rat of the hydraulic motor and with using set speed is (700) rpm.

SIMULATION ANALYSIS WITH AUTOMATION STUDIO V.52

(EHSS) for speed control of hydraulic motor has been built and simulated by using Automation Studio (AS) package V5.2 . This package was developed in (2003) by (FAMIC Technologies inc / Canada) to contain comprehensive libraries of hydraulic, pneumatic, ladder logic, and digital electronic symbols . This package is completely integrated software package that allows the user to design, simulate and animate circuits consisting of various automation technologies. After complete the design of (EHSS) for speed control, simulation process can be done. The simulation process is shown in **fig. 10**

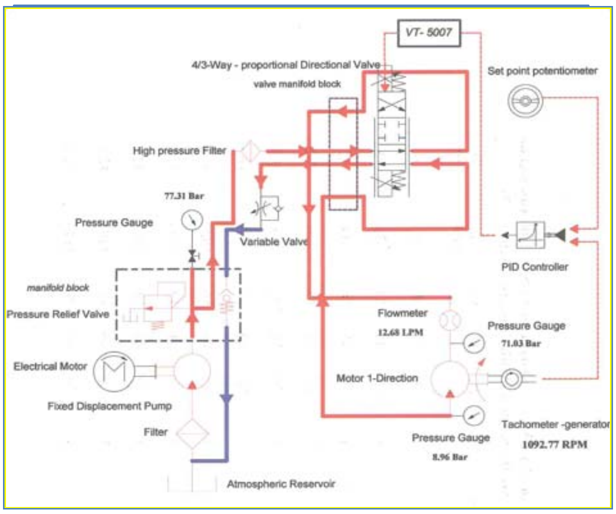

Fig (10): Simulation process of (EHSS) using (AS)

Fig. 11 shows the plotter window of this simulation process. (Automation Studio 2008)

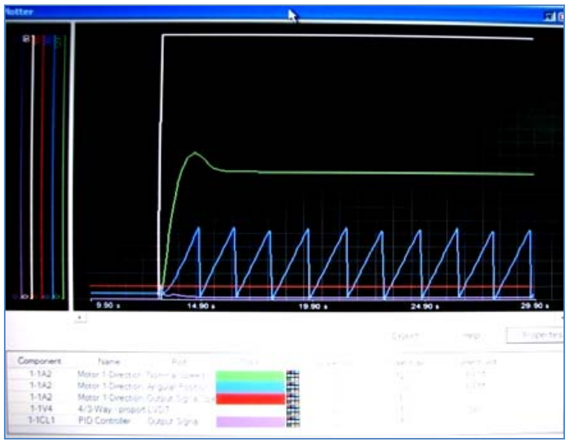

Fig (11): The plotter window for (AS)

RESULTS AND DISCUSSION

The torque required to overcome the friction when the hydraulic motor started from the rest from first test . **Fig. 12** represent pressure drop with motor rotational speed using the conversion ($T = P\ D_m$) .The static friction is (1.92) N.m . The coulomb friction is (0.96) N.m . Viscous coefficients is (0.0051) N.m.sec/rad. The characteristic tests of the (Valve – motor) system has been done with two cases of supply pressure (70 , 40) bar. **Fig. 13** and **Fig. 14** show the relation between the input signal with rotation speed and flow rate .When the input signal value increases, the hydraulic motor velocity increases because increasing the input signal leads to increased volumetric flow rates and the hydraulic velocity as long as the increases. The difference between experimental and theoretical results due to losses in pressure in the system and friction effect and nonlinearity in system. if the supply pressure increased ; volumetric flow rate increased as well as the hydraulic motor speed. And plotting experimentally the variation of output voltage with input voltage to the proportional amplifier VT-5007. The slop of this curve is the gain $K_{amp.}$ of VT - 5007 and equal to (0.93) .

Fig. 15 shows the velocity - form of hydraulic motor under open loop system from third test of EHSS as a oscilloscope window represented the relation between the output voltage (mille V) of tachogenerator and the time (sec.).this figure show the rise time and steady state and transient region .

Figs. 16 shows the velocity - form of hydraulic motor under close - loop system from third test of (EHSS) as a oscilloscope window .with best (PID) gains using two methods, trail and error methods and Ziegler and Nichols methods. **Table 1** shows the difference in PID gain between experimental and simulation analysis.

Fig.17 and **Fig. 18** show the speed control tests results of the oscilloscope windows of (EHSS) under 65 ± 5 and 45 ± 5 C° of hydraulic oil temperature under load disturbance applied of the hydraulic shift ,experimentally the performance of (EHSS) of speed control is best at 65 ± 5 C° and give smooth response under load disturbance and short time to return to set speed . **Table 2** and **Table 3** show the speed control parameters test with hydraulic temperature at 65 ± 5 and 45 ± 5 C° respectively. Generally it is clear that this system has good response as it reaches the final value smoothly without any overshoot.

Fig. 19 represented comparison between the PID controller methods to calculate the gains at simulation process by (AS) package V5.2 , and the best (PID) gains for speed controller of EHSS clear that the simulation gains are approximately same the experimental work.

Fig. 20 shows the simulation process of (EHSS) of speed control using AS package V5.2 under same experimental applied load toque on hydraulic shift, it is clear that using simulation analysis is agree with experimental work expected small delay , the experimental results are represented the real behavior of the system under many nonlinearity factors like frictions and compressibility while the simulation results represented the simulation process for the system depended on input data for all components in the simulation program.

Figs. 21 show the step response of (EHSS) for speed control by using MATLAB package V7.1 simulation analysis. It is clear that step response is agree with experimental and simulation analysis. **Table 4** shows Self - turn parameters for (EHSS) by MATLAB package V7.1 ,and show the stability of (EHSS) , approximately, all the gains of the (PID) controller form these methods are same value.

Figs. 22 shows the bode diagram of (EHSS) ,from the bode diagram, the peak gain margin is 1.76 db at phase margin is 145°, all meet system stabilizing requirements and good dynamic quality can be secured.

Table (1): (PID) controller gains methods for (EHSS)

(PID) gains calculated methods	K_p	K_i	K_d
Trial and error method (experimentally)	3	1-10	0-0.5
Ziegler and Nichols method (experimentally)	2.04	5.1	0.2
Trial and error method (simulation analysis)	3	5	0.2
Ziegler and Nichols method (simulation analysis)	2.1	4.2	0.26
Self- turn parameters by MATLAB package V7.1	2.6	6.3	0.1

Table (2): Speed control parameter test with Hydraulic oil temperature 65 ± 5°C

Torque N.m	Set ω_m rpm	Min. ω_m rpm	Max. ω_m rpm	Time required sec.
0.38	700	690	715	1 0.8
0.74	703	686	724	1.3 1
1.5	702	652	780	1.5 1
2.8	705	632	855	1.8 1.2

Table (3): Speed control parameters test with Hydraulic oil temperature 45 $\pm 5°C$

Torque N.m	Set ω_m rpm	Min. ω_m rpm	Max. ω_m rpm	Time required sec.
0.38	702	685	720	1.5 1
0.74	708	674	733	2 1.3
1.5	705	645	805	2.2 1.5
2.8	708	622	864	2.5 1.8

Table (4) : Self turn parameters for (EHSS) using MATLAB package V7.1

Performance and Controller parameter	Turned value from MATLAB
Proportional gain (P)	2.6
Integral gain (I)	6.3
Derivative gain(D)	0.1
Rise time (sec.)	0.16
Gain margin (db at rad/sec.)	92 at 2.45e+003
Phase margin (deg at rad/sec.)	60.6 at 8.21
Closed - loop stability	Stable system

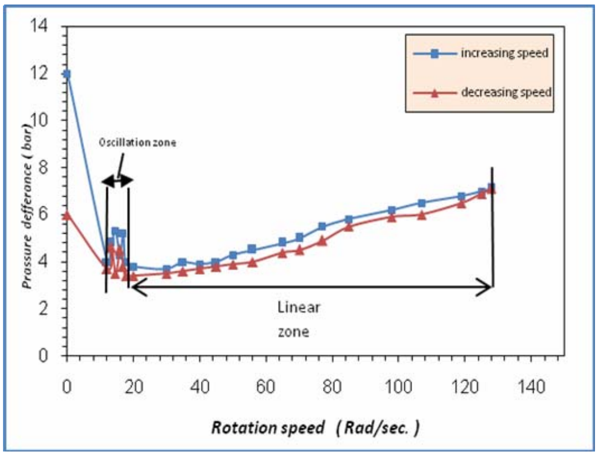

Fig (12) : Variation of rotation speed with pressure difference across the hydraulic motor

Table (5): Source of parameters of (EHSS) for speed control

Sym.	Description	Value	Unit	Source of Information
K_v	Proportional valve gain	$53*10^{-5}$	m/V	Manufacturer data
x_v	Spool stroke	$4.5*10^{-3}$	m	Manufacturer data
K_q	Valve flow gain	1.7	m^2/sec.	Calculated
K_c	Valve pressure gain	$2*10^{-9}$	m^3/sec. Pa	Calculated
d_{spool}	Spool diameter	$12*10^{-3}$	m	Manufacturer data
P_T	Return tank Pressure	$3*10^5$	N/m^2	From experimental
P_s	Supply pressure	$70*10^5$ $40*10^5$	N/m^2	From experimental
T_{fm}	Motor friction torque	2.92	N.m	Calculated experimentally
J_m	Inertia of motor and load	$32*10^{-5}$	$Kg.m^2$	Calculated
V_t	Total fluid volume in pipes	$24*10^{-5}$	m^3	Calculated
C_{tm}	leakage coefficient of motor	$1.5*10^{-13}$	m^3/sec. pa	Calculated experimentally
B_m	Viscous damping coefficient of motor	0.0051	N.m.sec./rad	Calculated experimentally
D_m	motor displacement	$16*10^{-7}$	m^3 /rad	Manufacturer data
β	Effective bulk modulus	$108*10^8$	N/m^2	Manufacturer data
ρ	Density of the Hydraulic oil	885	Kg/m^3	Manufacturer data
C_d	Discharge coefficient	0.65	---	Constant
D_p	Pump displacement	$16*10^{-7}$	m^3/ rad	Manufacturer data
ω_p	Pump rotational speed	146.5	rad/sec.	Manufacturer data
$K_{tach,}$	Tachogenerator gain	0.00264	V/rpm	Calculated experimentally

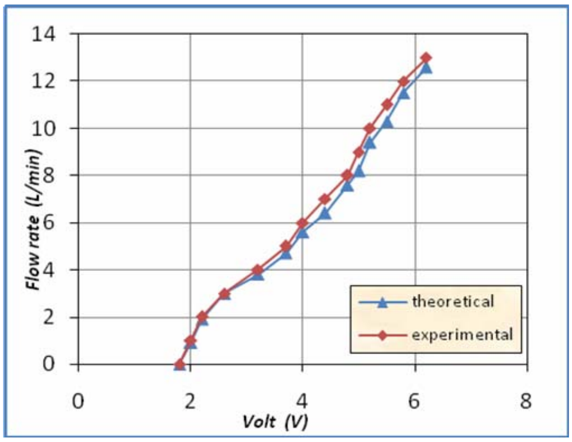

Fig (13) : Variation of hydraulic motor input flow rate with valve input signal (70 bar supply pressure)

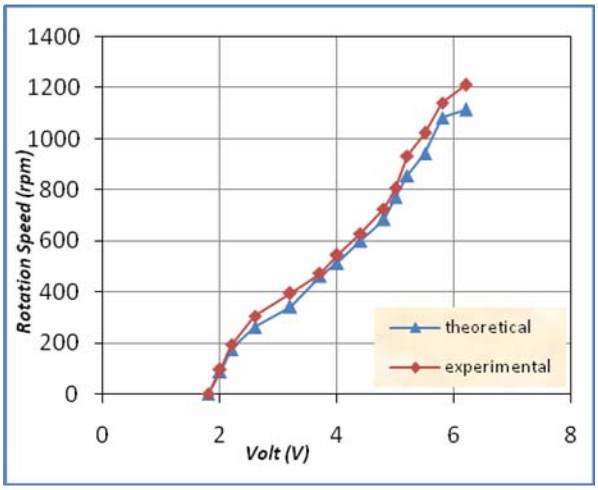

Fig (14): Variation of hydraulic motor rotation speed with valve input signal (70 bar supply pressure)

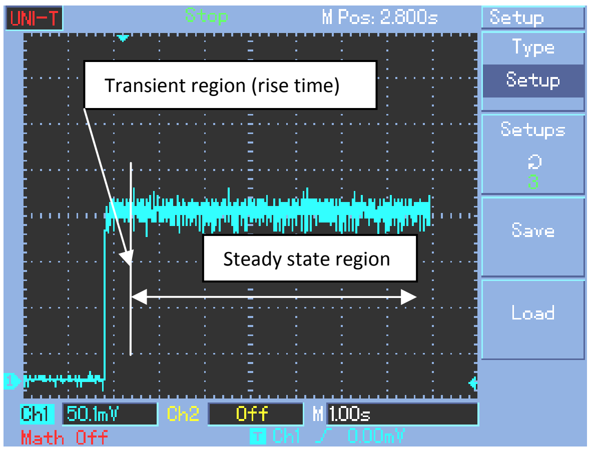

Fig (15) : Velocity of the hydraulic motor under open – loop system (70) bar

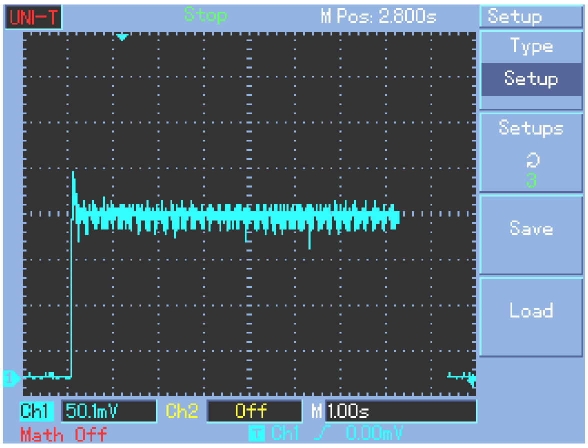

Fig (16): Velocity of the hydraulic motor under (PID) controller by Ziegler and Nichols method.

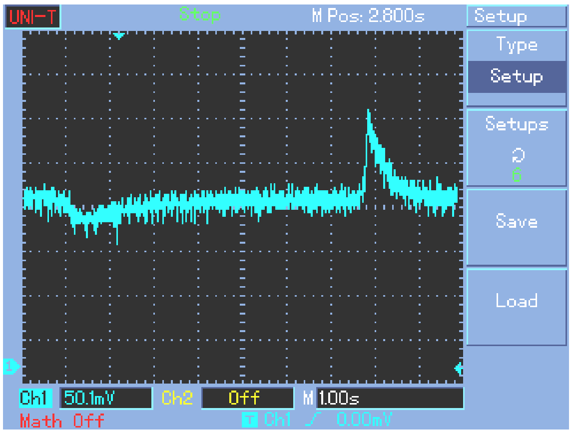

Fig (17) : Velocity of the hydraulic motor under (PID) controller with 2.8 N .m torque applied at 45 ±5 C° hydraulic oil temperature

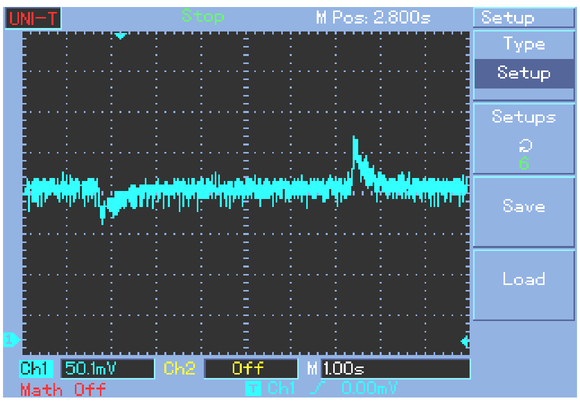

Fig (18) : Velocity of the hydraulic motor under (PID) controller with 2.8 N .m torque applied at 65 ±5 C° hydraulic oil temperature

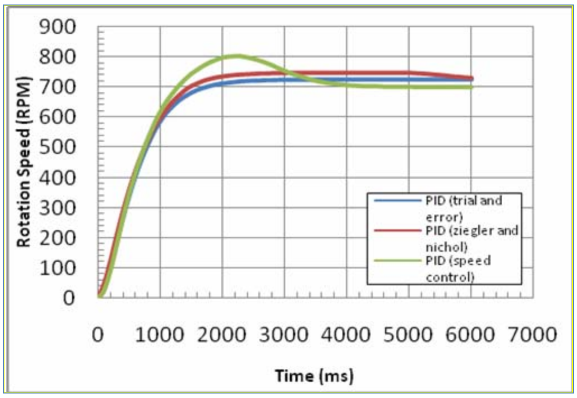

Fig (19): Simulation velocity of the hydraulic motor under (PID) controller by different methods

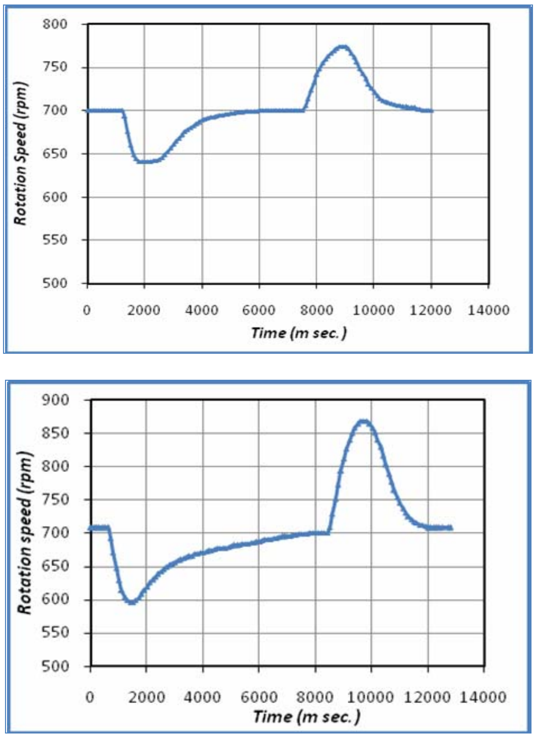

Fig (20) : Simulation velocity of hydraulic motor under (PID) controller with difference load

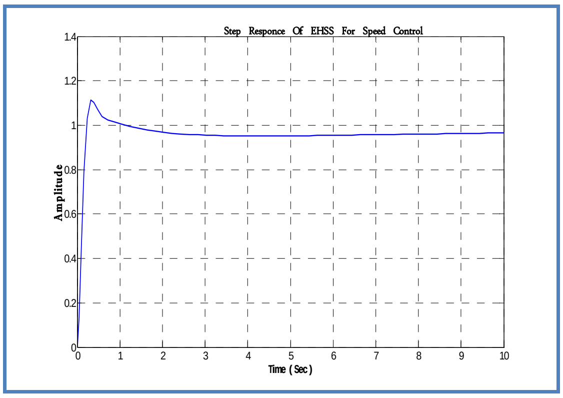

Fig (21): Step response of (EHSS) for speed control with MATLAB package V7.1

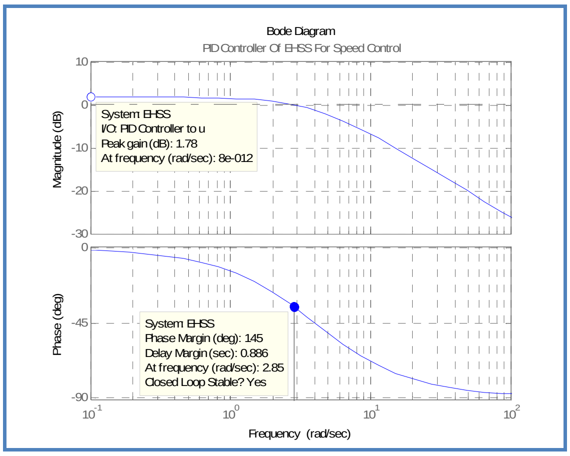

Fig (22) : Bode diagram for (no load speed TF) of (EHSS) for speed control

CONCLUSIONS

A model equations for (proportional valve – gear motor) combination system has been derived and verification, speed control system for hydraulic motor using simple low cost (PID) controller to get constant speed range of hydraulic motor under applied load .The range of speed control of the hydraulic motor depended on the system design like size of the hydraulic motor and the servo or proportional valve used, (flow rate) ,and reference speed.

Experimental tests showed that the performance and response of (EHSS) for speed control is best when the temperature of the hydraulic oil is **65** $\pm 5°C$ compared with low oil temperature oil **45** $\pm 5°C$. Experimental and simulation results when using (AS) program shows a convergence in the results and .This leads us to the possibility of using this program for testing and analysis and design of any hydraulic system and deal with two types of users (professional or beginner).

ACNOWLEDGMENT

We would like to thank very much to all the staff of mechanical engineering department of the Baghdad University for them cooperation and help.

REFERENCES

Aichao ; Xiangdong Z.; Gang L. .(2008)" Simulation Study on Constant Speed Output Control of Fixed Displacement Pump-Variable Displacement Motor Hydraulic System" Fluid Power and Mechatronics, IEEE Xplore. .PP 276 - 281.

Automation Studio (AS) user manual 2008, "Automation Studio circuit design simulation software", Fluid Power and Automation Technologies" , http://www. automation studio .com

Ambuel, J., Steenhoek, L., Smith, R., and Colvin, T.(1993) "Control of hydrostatic transmission output speed Development and comparison of PI and hybrid fuzzy-PI controllers", Transactions of the ASAE, Vol. 36, PP 1057 -1064.

Hossam. ,Mohammad El- Bardini (2011)"Implementation of speed controller for rotary hydraulic motor based on LS-SVM" journal of Expert system with applications . Vol.(38) PP.14249-14256.

Jacqueline ,Michael J.,Reza K. (2008) ," control engineering an introductory course " Palgrave ,New work ,USA.

Dasgupta, J. Watton ,S. Pan (2005) "Open-loop dynamic performance of a servo-valve controlled motor transmission system with pump loading using steady-state characteristics" journal of Mechanism and Machine Theory, Elsevier Ltd, Vol. (41) PP. 262-282.

Thrower , Kiefer S., Kelmer K., and Silverberg L. (1998) " Basic Experiments in PID Control for Non-electrical Engineers" Retrieved from http://www.mae.ncsu.edu/silverberg/piddocs.pdf.

Merritt , H. (1967) ," Hydraulic control systems", John Wiley and Sons Inc, New York ,US ,first edition.

Vladimir . (2006) "Design and modeling of a new electro hydraulic actuator " National Library of Canada, M.Sc thesis ,Mechanical engineering ,Toronto ,Canada.

Web site 1 "speed control for hydraulic motor" from http://www.hydraulicspneumatics.com.

Yasukazu, (2000)"Rotational speed control of hydraulic motort using sliding mode control" Journal of Japan society of mechanical engineering, No. (00-1579) , PP.2559-2564.

NOTATION

Sym.	Description	unit
A	Orifice area gradient	m^2
B_m	Viscous damping coefficient of hydraulic motor	N.m.sec/rad
C_{tm}	leakage coefficient of hydraulic motor	m^3/sec.Pa
d_{spool}	Spool diameter	m
D_m	Volume displacement rate of the hydraulic motor	m^3 /rad
J_m	Total Inertia of motor and load	Kg.m^2
K_p	Proportional gain	dimensionless
K_i	Integral gain	dimensionless
K_d	Derivative gain	dimensionless
K_c	Valve pressure gain	m^3/sec. Pa
K_v	Proportional valve gain	m/V
K_q	Valve flow gain	m^2/sec.
$K_{tach,}$	Tachogenerator gain	V/rpm
Δ	difference	
P_1 , P_2	Inlet and outlet pressure of the hydraulic motor	Pa
P_L	Load pressure	Pa
P_s	Supply pressure	Pa
P_o , P_T	Tank pressure	Pa
Q_1, Q_2	Input and output flow rate of the hydraulic motor	m^3/sec.
t	Time	sec.
T_s	Static friction torque	N.m
T	Total friction torque on the hydraulic motor	N.m
T_{fm}	Coulomb friction torque of the hydraulic motor	N.m
T_g	Torque generated by motor	N.m
T_m	External load torque on the hydraulic motor	N.m
$e(t)$	Input error signal of (PID)	V
u (t)	Output of (PID) signal	V
V_t	Total fluid volume in pipes	m^3
x_v	Spool valve position	m
ρ	Density of the Hydraulic oil	Kg/m^3
ν	Kinematic viscosity	mm^2/sec.
ω_h	Hydraulic natural frequency	rad /sec.
ω_m	Angular speed of the hydraulic motor shaft	rad /sec.
β	Effective bulk modulus	N/m^2
δ_h	Hydraulic damping coefficient	dimensionless

ABBREVIATION	MENE
AS	Automation studio
EHSS	Electro- hydraulic servo system
Sym.	Symbol
TF	Transfer function
PID	Proportional integral derivative
PV	Proportional valve